NASA Technical Paper 1134

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JANUARY 1978

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NASA
National Aeronautics
and Space Administration

Scientific and Technical
Information Office

1978

METHOD FOR CALCULATING CONVECTIVE HEAT-TRANSFER COEFFICIENTS OVER TURBINE VANE SURFACES

by Daniel J. Gauntner and James Sucec*

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SUMMARY

A method for calculating laminar, transitional, and turbulent convective heat-transfer coefficients over the surfaces of turbine vanes is described. Results of calculations with an approximate integral solution method are compared with results from a turbulent flat plate solution and a finite-difference solution.

The approximate integral solution method produced heat-transfer coefficients in good agreement with the finite-difference solution method in the laminar and fully turbulent flow regimes. The transitional flow regime length was approximately the same although the regime began earlier for the finite-difference solution. The turbulent flat plate solution produced a conservative heat-transfer distribution which exceeded the finite-difference solution by as much as 150 percent.

The calculations were made for given gas stream conditions of turbine inlet temperature equal to 1273 K, turbine inlet pressure of 45.7 newtons per square centimeter, a fuel/air ratio of 0.0, a vane channel inlet Mach number of 0.31, and an exit Mach number of 0.84. Transition criteria were based on momentum thickness Reynolds numbers in the integral solution.

INTRODUCTION

In the design of blades and vanes for gas turbine engines, the convective heat-transfer rate from the gas stream to the surface must be calculated. The simplest approach is to use the flat plate solution. The flat plate solution, however, applies only to zero pressure gradient flow. In many instances, though, this approximation leads to

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satisfactory results for initial calculations because the Stanton number is relatively insensitive to pressure gradient (ref 1). One of the most powerful methods of calculating the heat flux to a turbine blade is to solve the boundary layer mass, momentum, energy, and turbulent energy equations simultaneously by a finite-difference approach (ref. 2). In between these two approaches is the approximate integral solution. This method is more general than the flat plate solution, but not as powerful as the finite-difference methods.

An approximate integral solution (ref. 3) for the local surface heat-transfer coefficient that takes into account the effect of pressure gradient and of variable surface temperature is reviewed in this report. The turbulent and transitional heat-transfer coefficients are based on the approximate solutions of the equations for the thermal boundary layer. The laminar coefficients are those due to reference 4 with the corrections for variable wall temperature due to reference 5. The boundary layer transition criteria used in determining the heat-transfer coefficients are based on the boundary layer momentum thickness Reynolds number. Reference 6 discusses data from the literature and presents an empirical expression relating the start of transition to the critical momentum thickness Reynolds number. These Reynolds numbers are presented as a function of a gas stream pressure gradient parameter and the gas stream turbulence intensity level.

SYMBOLS

A_o	constant
B	constant, $A_o Pr^{-n}$
b	variable used in eq. (5)
C_p	specific heat
D	diameter of leading edge of vane
Eu	Euler number, $-(dP_g/dx)/(\rho_g u_s^2/x)$
$F_{\ell am}$	$Nu/Re_x^{0.5}$
h_g	heat-transfer coefficient
k	thermal conductivity
m, n	exponents
Nu	Nusselt number
P	pressure

Pr	Prandtl number
R	radius of leading edge
Re	Reynolds number
St_x	local Stanton number, $h_{g_x}/\rho_s C_p u_s$
T	temperature
T_*	temperature arbitrarily taken from vane temperature distribution
Tu	turbulence level
$\overline{Tu}$	mean turbulence level
u_s	local free-stream velocity
V_{cr}	local critical velocity
x	local distance from leading edge of vane
δ_T^*	enthalpy thickness
θ	momentum thickness
λ	pressure gradient parameter
μ	dynamic viscosity
ν	kinematic viscosity
ρ	density

Subscripts:

crit	critical location of flow transition
ge	effective gas
g_x	local gas
$g_{(eq.\ 3)}$	gas condition for eq. (3)
ℓam	laminar flow conditions
m	metal
rat	correction factor on constant surface temperature coefficient
ref	reference
s	static
T_i	turbine inlet
t	turbulent

w wall

x local condition

0 refers to positions where flow ceases to be laminar or where it becomes fully turbulent

ANALYSIS

The design of turbine vanes and blades for gas turbine engines requires accurate convective heat-transfer coefficients around the entire airfoil surface. Around the exterior cylindrical leading edge of the vane, the cylinder correlation is commonly used. For the remaining surfaces of the vane, laminar, transition, and turbulent coefficients must be supplied.

Leading Edge Region

Around the leading edge of a turbine vane the heat-transfer correlation for a cylinder in cross flow (ref. 7) gives

$$h_g = 1.14\ Re^{0.5} Pr^{0.4} \frac{k}{D}\left[1 - \left(\frac{|\theta|}{90}\right)^3\right] \qquad 0 < |\theta| < 80^\circ \tag{1}$$

where $Re = \rho u_g D/\mu$ and ρ is determined from the perfect gas law. All properties for the cylinder correlation are evaluated using air properties at the reference temperature (ref. 8):

$$T_{ref} = 0.5\ T_m + 0.28\ T_s + 0.22\ T_{ge} \tag{2}$$

Recent tests have shown that leading edge turbulence could increase the heat-transfer coefficient by a factor of up to 1.8 (ref. 9). This effect can be estimated (ref. 10) if the turbulence characteristics are known. The correction for this effect, called an augmentation factor, acts as a factor multiplying equation (1).

Laminar Flow Region

For the laminar flow region, the local convective heat-transfer coefficients are obtained from previous results (ref. 4):

$$h_{g_x} = \overline{F}_{\ell am} Re_x^{0.5} Pr^{1/3} \left[\frac{k}{x}\right] \tag{3}$$

where

$$\overline{F}_{\ell am} = \frac{Nu}{Pr^{1/3} Re_x^{0.5}}$$

$$Re_x = \frac{\rho u_s x}{\mu}$$

The $\overline{F}_{\ell am} Pr^{1/3} = F_{\ell am}$ is evaluated as a function of the local Euler number Eu and the local ratio of T_s/T_m. The values of $F_{\ell am}$, or $Nu/Re_x^{0.5}$, are approximated by solutions for wedge-type flow. These wedge solutions may be used to approximate heat-transfer coefficients along an arbitrary profile. The value of Eu at any position along the profile determines the corresponding wedge for which, at the same distance x from the stagnation point, the heat transfer on the wedge and the arbitrary profile are assumed to be equal.

The convective heat-transfer coefficients obtained from equation (3) are based on constant wall temperature. Since turbine blades have varying wall temperatures, it is necessary to correct the convective coefficients to account for such variations. A method for making such a correction exists (ref. 5). The method determines the ratio of the variable surface temperature coefficient to the constant surface temperature coefficient:

$$h_g = h_{rat} h_g \text{ (eq. (3))} \tag{4}$$

where h_{rat} is a function of b and b is determined from

$$\frac{T_{ge} - T_m}{T_{ge} - T_*} \propto x^b$$

and T_m is the local surface temperature, T_* is a reference wall temperature, and x is the distance from the stagnation point. One set of data of h_{rat} against b has been curve fit as follows (ref. 5):

$$h_{rat} = -0.111\, b^2 + 0.318\, b + 1.0 \qquad 0 \le b \le 0.6 \tag{5a}$$

$$h_{rat} = -0.520\, b^2 + 0.310\, b + 1.0 \qquad -0.5 \le b < 0 \tag{5b}$$

$$h_{rat} = -4.977\, b^2 - 4.288\, b - 0.186 \qquad -0.76 \le b < -0.5 \tag{5c}$$

Turbulent Flow Region

The technique for predicting turbulent local convective heat-transfer coefficients is based on the approximate integral solution (ref. 3). The solution of the equation for the thermal boundary layer takes into approximate account the effects of thermal history, free stream velocity variation, and free stream density variation.

In the solution, the integral energy equation was arranged into a form involving the local enthalpy thickness δ_T^*:

$$\frac{d\delta_T^*}{dx} + \left[\frac{1}{\rho_s}\frac{d\rho_s}{dx} + \frac{1}{u_s}\frac{du_s}{dx} + \frac{1}{(T_w - T_s)}\frac{d(T_w - T_s)}{dx}\right]\delta_T^* = \frac{h_x}{\rho_s C_p u_s} \tag{6}$$

If a relation could be found between $St_x = h_x/\rho_s C_p u_s$ and δ_T^*, equation (6) could be solved for δ_T^* as a function of x and therefore h_x as a function of x. For a zero pressure gradient, constant property flow over an isothermal flat plate, the Stanton number has the following form:

$$St_x = A_0 Re_x^{-m} Pr^{-n} \tag{7}$$

The values of A_0, m, and n depend on whether the flow is laminar, transitional, or turbulent. Substituting equation (7) into equation (6) gives the functional form of the Stanton number for the flat plate,

$$St_x = B^{1/(1-m)}(1 - m)^{-m/(1-m)} Re_{\delta_T^*}^{-m/(1-m)} \tag{8}$$

where $B = A_0 Pr^{-n}$. Furthermore, the St_x in equation (6) was proposed to be some general function of $Re_{\delta_T^*}$ but otherwise independent of body shape, surface temperature variation, free-stream velocity, and density variation. If this is true, a universal relationship between St_x and $Re_{\delta_T^*}$ must be given by equation (8). If equation (8) is substituted into equation (6) and the resulting expression integrated, the following result will be applicable:

$$\frac{h_x}{\rho_s C_p u_s} = \frac{B^{1/(1-m)}(1-m)^{-m/(1-m)}\left[\mu_s(T_s - T_w)\right]^{m/(1-m)}}{\left\{B^{1/(1-m)}(1-m)^{1/(m-1)}\int_{x_0}^{x}\frac{\rho_s u_s(T_s - T_w)^{1/(1-m)}}{\mu_s^{m/(m-1)}}\,dx + \left[\rho_s u_s(T_s - T_w)\delta_T^*\right]_{x_0}^{1/(1-m)}\right\}^m} \tag{9}$$

where $x_0 = x_{crit}$.

To evaluate B and m in equation (9), the expression for the turbulent local convective heat-transfer coefficient for a flat plate is used; namely,

$$h_{g_x} = 0.0296\ Re_x^{0.8} Pr^{1/3}\frac{k}{x} \tag{10}$$

Expressing equation (10) in the form of equation (8) gives $B = 0.0296\ Pr^{-2/3}$ and $m = 0.2$. Assuming the free-stream dynamic viscosity μ_s is only a weak function of x,

$$\frac{h_{g_x}}{\rho_s C_p u_s} = \frac{0.0296\ Pr^{-2/3}(T_s - T_w)^{0.25}\left(\frac{T_s}{T_w}\right)^{0.25}}{\left\{\int_{x_0}^{x}\frac{\rho_s u_s(T_s - T_w)^{1.25}}{\mu_s}\,dx + \left[\frac{0.8}{0.0296\ Pr^{-2/3}}\left(\frac{\rho_s u_s \delta_T^*}{\mu_s}\right)(T_s - T_w)\right]_{x_0}^{1.25}\right\}^{0.2}} \tag{11}$$

The factor $(T_s/T_w)^{0.25}$ in the numerator of the right side of equation (11) is the temperature ratio correction for property variation across the boundary layer (ref. 11) for $T_w/T_s < 1$. The temperature-dependent fluid properties, then, are evaluated at the local free-stream static temperature.

Transition Flow Region

Over some portions of the surface of a turbine vane, the gas stream flow fluctuates intermittently between laminar and turbulent flow. This region of transition is not really described accurately by either equation (4) or equation (11). Thus, an empirical expression analogous to the form of equation (10) is sought which is applicable in the tran-

sition region. Experimental results for heat transfer in the transition region along a flat plate have been obtained (ref. 3). For Reynolds numbers $Re_x > 200\ 000$ the heat-transfer correlation in the transition region had the form

$$Nu_x = 0.000386\ Re_x^{10/9} \tag{12}$$

Assuming again the form of equation (8), with $Pr = 0.7$, the values $B = 0.000435\ Pr^{-2/3}$ and $m = -1/9$ result. When these values of B and m are substituted into equation (9), an expression of h_{g_x} as a function of x in the transition region results:

$$\frac{h_{g_x}}{\rho_s C_p u_s} = \frac{0.000435\ Pr^{-2/3} (T_s - T_w)^{-0.1}\left(\frac{T_s}{T_w}\right)^{0.25}}{\left\{\int_{x_0}^{x} \frac{\rho_s u_s (T_s - T_w)^{0.9}}{\mu_s} dx + \left[\frac{1.111}{0.000435\ Pr^{-2/3}}\left(\frac{\rho_s u_s \delta_\tau^*}{\mu_s}\right)(T_s - T_w)\right]_{x_0}^{0.9}\right\}^{-0.111}} \tag{13}$$

To use equations (11) and (13) it is necessary to evaluate δ_τ^* at x_0. Since δ_τ^* is continuous at x_0 (between laminar and transitional and between transitional and turbulent) it is possible to evaluate δ_τ^* by rearranging equation (8):

$$\left(\frac{\rho_s u_s \delta_\tau^*}{\mu_s}\right)_{x_0} = \frac{B^{1/m}}{1 - m}\left(\frac{h_{g_x}}{\rho_s C_p u_s}\right)_{x_0}^{1-\frac{1}{m}} \tag{14}$$

At x_0, the values of m, B, and $h_{g_x}\big|_{x_0}$ are from the laminar and transitional expressions. They are used in equations (13) and (11), respectively. For equation (11), $B = 0.000435\ Pr^{-2/3}$ and $m = -1/9$. For equation (13), $B = 0.332\ Pr^{-2/3}$ and $m = 0.5$ when $\overline{F}_{lam} = 0.332$ in equation (3).

Equations (11) and (13) are finally rewritten in useful form giving for the turbulent h_{g_x}

$$
h_{g_x} = \frac{\rho_s C_p u_s \,(0.0296)\, Pr^{-2/3} \,(T_s - T_w)^{0.25} \left(\dfrac{T_s}{T_w}\right)^{0.25}}{\left\{ \displaystyle\int_{x_0}^{x} \frac{\rho_s u_s (T_s - T_w)^{1.25}}{\mu_s}\, dx + \left[\frac{0.8\,(0.9)\,(0.000435)}{0.0296} \left(\frac{\dfrac{h_{g_x}}{\rho_s C_p u_s}}{0.000435\, Pr^{-2/3}} \right)^{10} (T_s - T_w) \right]_{x_0}^{1.25} \right\}^{0.2}} \tag{15}
$$

and for the transitional h_{g_x}

$$
h_{g_x} = \frac{\rho_s C_p u_s \,(0.000435)\, Pr^{-2/3} \,(T_s - T_w)^{-0.1} \left(\dfrac{T_s}{T_w}\right)^{0.25}}{\left\{ \displaystyle\int_{x_0}^{x} \frac{\rho_s u_s (T_s - T_w)^{0.9}}{\mu_s}\, dx + \left[\frac{1.111\,(2)\,(0.332)^2\, Pr^{-2/3}}{0.000435 \left(\dfrac{h_{g_x}}{\rho_s C_p u_s} \right)} (T_s - T_w) \right]_{x_0}^{0.9} \right\}^{-0.111}} \tag{16}
$$

Transition Criteria

The change from laminar to transitional flow and from transitional to turbulent flow occurs when the Reynolds number becomes sufficiently high and thus allows instabilities in the boundary layer to grow. To account for local variations in the boundary layer, a Reynolds number based on the momentum thickness is used to determine the values of x_0 in equations (15) and (16) where changes between flow regimes occur. An approximate expression for the momentum thickness can be used (ref. 12):

$$\theta = \frac{0.67\ \mu_s^{0.5}}{\rho_s^{0.5}\ u_s^3}\left(\int_0^x u_s^5\, dx\right)^{0.5} \tag{17}$$

Thus, the Reynolds number based on momentum thickness can be found as a function of x:

$$Re_\theta = \frac{\rho_s u_s}{\mu_s}\left[\frac{0.67\ \mu_s^{0.5}}{\rho_s^{0.5}\ u_s^3}\left(\int_0^x u_s^5\, dx\right)^{0.5}\right] \tag{18}$$

The values of x_0 (where changes occur from laminar to transitional flow) can then be found from equation (18). This equation is assumed to apply also in the transitional regime and up to the location where the flow goes from the transitional to turbulent. The question arises as to what to use for $Re_{\theta\, x_0}$. Empirical expressions are given (ref. 6) for the value of Re_θ at the start of transition (that is, the change from laminar to transitional flow) for a zero pressure gradient case and a nonzero pressure gradient case. They are, respectively,

$$Re_\theta = 190 + \exp\,(6.88 - 1.03\ Tu) \tag{19}$$

$$Re_\theta = [0.27 + 0.73\exp\,(-0.8\ \overline{Tu})]\ [550 + 680\ (1 + \overline{Tu} - 21\lambda)^{-1}] \tag{20}$$

where

$$\lambda = \frac{\theta^2}{\nu}\frac{du_s}{dx}$$

$$Tu = \frac{\text{rms velocity fluctuation} \times 100}{u_s}$$

and the mean turbulence level $\overline{Tu}$ characterizes the flow throughout the history of the boundary layer. For $\lambda = 0$, equation (20) agrees with equation (19) within ±5 percent. Very little information exists as to what value of Re_θ to use when the change from transition to turbulent flow occurs, but a value of $Re_\theta = 360$ is used (ref. 11).

RESULTS AND DISCUSSION

A method which calculates laminar, transitional, and turbulent convective heat-transfer coefficients over the surfaces of turbine vanes is presented. A sample calculation was made for the suction surface of a J-75 size turbine vane whose profile is identical to that given in reference 13. The calculation was made for given gas stream conditions of turbine inlet temperature equal to 1273 K, turbine inlet pressure of 45.7 newtons per square centimeter, fuel/air ratio of 0.0, vane channel inlet Mach number of 0.31, and a vane channel outlet Mach number of 0.84. The vane metal temperature and free-stream dimensionless velocity distributions used in the calculation are given in figure 1. The gas stream conditions given previously were used in the quasi-three-dimensional computer program of reference 14 to calculate the velocities. A short computer program was written to perform the calculation of the heat-transfer coefficients. These velocities were input directly into this program.

The calculated convective heat transfer coefficient distribution for the vane suction surface is shown in figure 2. Also shown in the figure are the distributions obtained from the turbulent flat plate equation and from a finite-difference computer program (ref. 2). The properties of air were used in all cases.

The flat plate distribution is for a flow assumed to be laminar over the cylindrical leading edge and completely turbulent over the remaining vane surface. While closer agreement would be obtained by using the transition criteria with the gas stream velocity constant, the assumption shows the difference between the approximate integral solution and the more conservative (all turbulent) flat plate solution. The approximate integral solution and the finite-difference solution both show distinct regions of laminar, transitional, and turbulent flow.

The value of the momentum thickness Reynolds number for transition from laminar to transitional and from transitional to fully turbulent flow was assumed to be 200 and 360, respectively, for the integral solution. As a second case of the integral solution, the flow was assumed to be fully turbulent near 4.2 centimeters. This position corresponds to a film-cooling slot on the turbine vane profile of reference 13. The assumption of fully turbulent flow was based on the results of reference 15. Experimental temperature data were presented showing a film cooling layer causing a laminar or transitional flow to become turbulent. The finite-difference solution requires no a priori assumption regarding transition. The turbulent viscosity and thermal diffusivity are calculated from the turbulence model (turbulent kinetic energy and mixing length) and added to their respective molecular values. For transitional boundary layers, the turbulent and molecular components are the same order of magnitude.

For purposes of comparison assume that the finite-difference solution gives the baseline heat-transfer coefficient distribution. The turbulent flat plate solution (the conservative solution), except for the vane surface between 4.0 and 6.4 centimeters where

it is less than 10 percent different, is always 50 to 150 percent higher than the baseline. If laminar flow had been assumed at the beginning of the vane, the solution would have been lower than the baseline consistently. The approximate integral solution method, which accounts for the laminar, transitional, and turbulent flow regimes, produced a distribution which is within 15 percent of the baseline between 0.6 and 2.2 centimeters, where both programs predict that laminar flow exists. Between 2.2 and 4.2 centimeters, the integral solution returns transition flow heat-transfer coefficients an average 50 percent higher than baseline. The integral solution which is fully turbulent at the film cooling slot agrees with the baseline within 15 percent from 4.2 to 6.4 centimeters. The other integral solution does not become fully turbulent until 5.2 centimeters. From that point until 6.4 centimeters, the two integral solutions are in essential agreement. The baseline finite-difference solution goes to fully turbulent flow at 4.2 centimeters. The programmed solution accounted for a film-cooling blowing rate in its calculations. The surface region between 0.5 and 5.2 centimeters includes the two points of change (laminar to transitional and transitional to fully turbulent) and the transitional flow regime itself. The use of the laminar expression (eq. (18)) for the momentum thickness Reynolds number in the approximate integral solutions produces an early change to transitional flow and a later change to turbulent flow than is produced by the finite-difference solution, which uses the calculated boundary layer momentum thickness to arrive at the local momentum thickness Reynolds number. The expression for the transitional regime heat-transfer coefficient (eq. 16), while being of the same level, does not follow the finer details of the heat-transfer coefficient produced by the finite-difference solution near 1.0 centimeter. Figure 2 does imply that the heat-transfer rate to the vane is about equal for the integral and finite-difference solutions.

One possible source of error is the assumption that equation (18) is valid in transitional flow region and up to a value of 360. No assumption at all is made about the accuracy of Re_θ as calculated by equation (18) once the value exceeds 360. In addition, no provision for retransition is made.

For the variable surface temperature correction of equations (5a) to (5c), if b is greater than 0.6 or less than -0.76 it is set equal to the respective limit and h_{rat} is calculated. For the distribution of figure 2, all values of b were within the specified range. The correction for approximating wedge flow solutions is made according to equation (3). The correction is a function of Eu and T_s/T_m. If the interpolation for F_{lam} exceeds the range of tabulated values, F_{lam} is set equal to 0.332. Equation (3) then degenerates to the traditional laminar flat plate solution. In figure 2 this degeneration occurred at $x = 0.8$ centimeter.

The relative value of the approximate integral solution over the turbulent flat plate or the finite-difference solution lies with the user. For the user interested in a conservative order of magnitude value of heat-transfer coefficient, the turbulent flat plate solution and a desk calculator will satisfy his needs. The user of the finite-difference

solution program has the greatest flexibility; he will be able to solve the most difficult problems, such as the prediction of the local surface coefficient of heat transfer near film cooling slots and on the surface of full coverage film cooled vanes. The approximate integral solution method gives the user a heat-transfer-coefficient distribution which takes into approximate account the laminar, transitional, and turbulent flow regions. In situations which do not involve blowing into the boundary layer or severe compressibility effects, the user can obtain heat-transfer-coefficient distributions of acceptable accuracy with relatively little effort and with no need to deal explicitly with turbulent transport parameters. In addition, when adequate computer facilities are not available, the integral technique can be adapted to desk calculators, and it is even feasible for slide rule calculations.

SUMMARY OF RESULTS

An approximate integral solution method for calculating laminar, transitional, and turbulent convective heat-transfer coefficients over the surfaces of turbine vanes is described. An example calculation of heat-transfer coefficients using a turbulent flat plate solution, the integral solution, and a finite-difference solution was made for a turbine vane. The results are as follows:

(1) The approximate integral solution and the finite-difference solution agree well in the laminar and turbulent regions.

(2) The finite-difference solution predicts a transition region whose length approximates that predicted by the integral solution. Its onset begins much sooner, however.

(3) Use of either the integral solution or the finite-difference solution in a heat flux calculation would result in substantially lower levels of heat flux to the vane than would the use of the turbulent flat plate solution.

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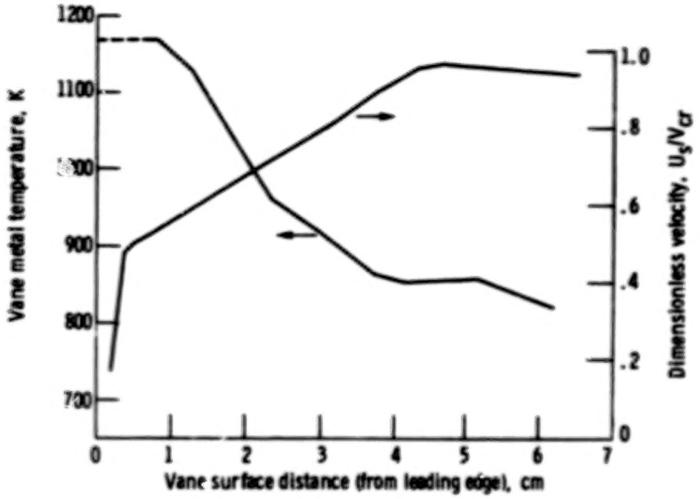

Figure 1. - Vane metal temperature and dimensionless velocity profiles.

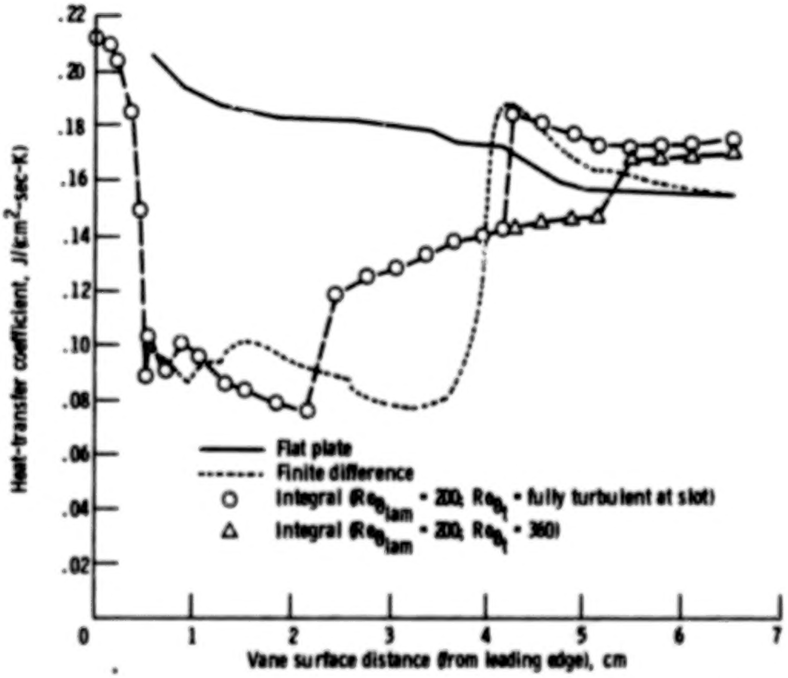

Figure 2. - Comparisons of heat-transfer coefficients using properties of air.

1. Report No. NASA TP-1134	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle METHOD FOR CALCULATING CONVECTIVE HEAT-TRANSFER COEFFICIENTS OVER TURBINE VANE SURFACES		5. Report Date January 1978
		6. Performing Organization Code
7. Author(s) Daniel J. Gauntner and James Sucec		8. Performing Organization Report No. E-9324
		10. Work Unit No.
9. Performing Organization Name and Address National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Paper
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

A method for calculating laminar, transitional, and turbulent convective heat-transfer coefficients for turbine vane surfaces is described. An approximate integral solution method produced results in good agreement with a finite-difference solution. Comparisons between the two are presented herein. The integral solution results agreed well with the finite-difference solution results in the laminar and turbulent regions. Differences in calculating the start of transition produced a later starting point for the approximate integral solution's transitional flow regime.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Heat transfer coefficients; Convective heat transfer; Approximate boundary layer solutions; Variable surface temperature; Variable pressure gradient	Unclassified - unlimited STAR Category 45

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	16	A02

* For sale by the National Technical Information Service, Springfield, Virginia 22161